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FLUID FILM LUBRICATED BEARING PRACTICE.

By G. S. ELFER, M.A. (Cantab.), A.M.I.Mech.E.

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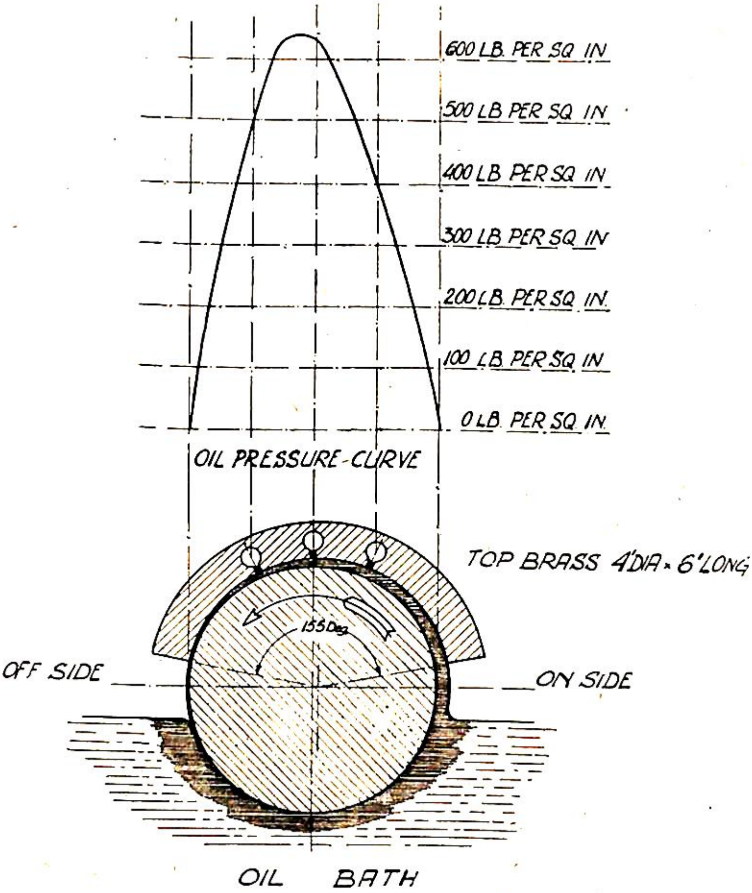


Fig. 1—B. Tower's Experiment.

Since S. Z. de Ferranti, thrust bearings with fixed tapers have been used. Like the Michell type thrust bearings, the theory of using converging surfaces between a moving and a stationary member to produce and maintain an oil film with load bearing capacity is the basis upon which the tapered-land thrust bearings are designed.

Following Reynolds, the hydrodynamic theory of film lubrication has been further extended by Sommerfeld, Gümbel, Goodman, Howarth and Swift, to mention some. These and others have advanced theory, by tending to eliminate the less rational assumptions and have substantiated their findings by experiments.

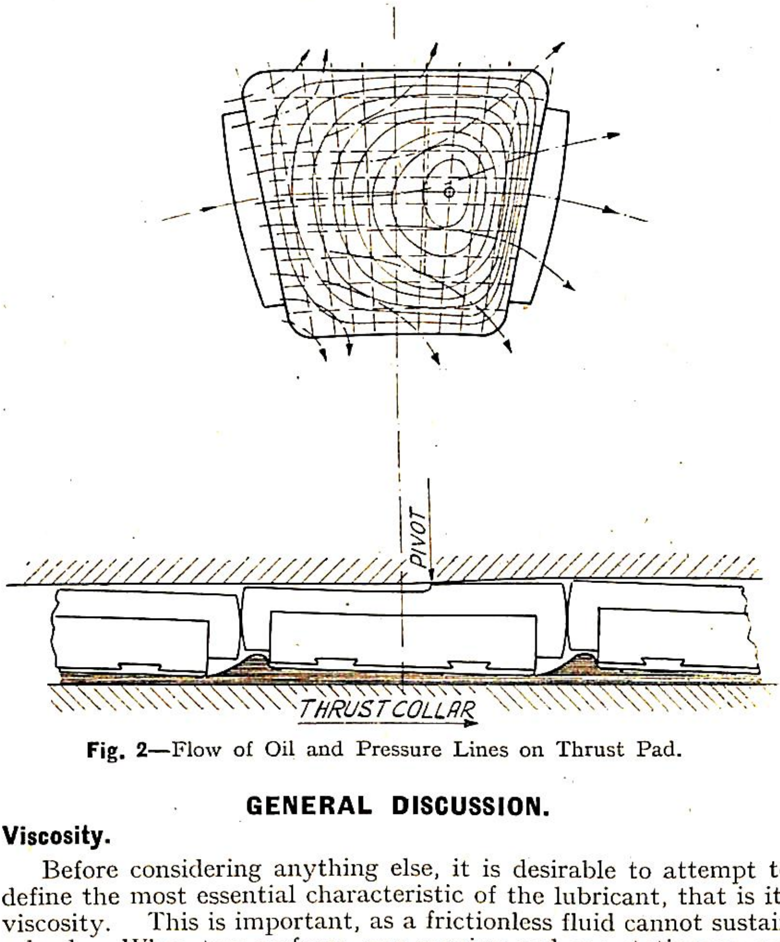


Fig. 2—Flow of Oil and Pressure Lines on Thrust Pad.

GENERAL DISCUSSION.

Viscosity.

Before considering anything else, it is desirable to attempt to define the most essential characteristic of the lubricant, that is its viscosity. This is important, as a frictionless fluid cannot sustain a load. When two surfaces, one moving and one stationary, are separated by oil, a thin layer of oil will adhere to both the stationary and the moving surface. As successive layers will move with velocities decreasing as the distance from the respective surfaces increases, this will give rise to a shearing between adjacent layers. The resistance of the fluid to this shear is measured by its viscosity. Hervey has illustrated the laminar flow with the aid of a thick book, by moving one of the covers relative to the other.

Film Lubrication Condition in Thrust Bearings.

The necessary condition for the establishment of a pressure film is that the thickness of the film should be greater at the inlet than at exit. If two surfaces are parallel, one sliding under another, the velocity of the subsequent layers will be represented by a right-angled triangle, where the base will represent the velocity of the moving surface. In such a case, if a load is applied to the upper surface, the fluid will be squeezed out and the surfaces will come into contact, unless the pressure at which the oil is applied is greater than the load carried. But when the upper surface is inclined, the film will sustain a load. If side leakage is neglected, the same quantity of fluid will enter at the leading edge of the moving surface as will leave at the trailing edge. The velocities of layers will be as represented by Fig. 3. The oil is dragged in and is crowded into an ever-narrowing space. To escape, the oil must leak either at the sides or along the leading edge; this is represented by the negative velocity loop.

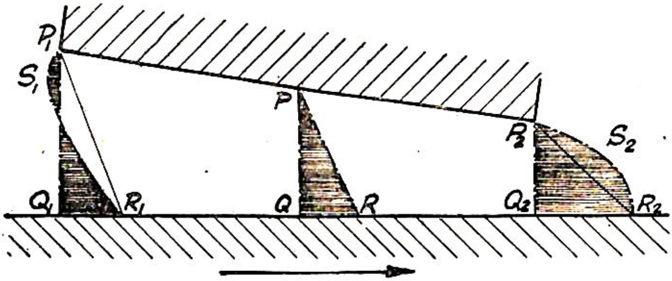


DIAGRAM SHOWING VELOCITY OF VISCOUS FLOW BETWEEN INCLINED SURFACES

Side Leakage Neglected

Fig. 3.

Therefore, the size of load P carried depends on—

(*a*) the velocity of the moving surface, N,
(*b*) the shape and dimensions of the bearing surface,
(*c*) the viscosity of the fluid, Z.

Equilibrium is determined by the parameter ZN/P and by the minimum film thickness h; the latter also governs the factor of safety of the bearing. On the other hand, the factor governing the incidence of the stationary surface, in relation to the moving surface, will be the pivot axis through which the load is transmitted to the fluid film. This axis should, therefore, not be colinear with the resultant fluid pressure. Theory indicates that the position of support of the pressure surface for the least friction is in the region of 0·4 times the length measured along the mean and from the exit edge. The thickness of the film at the trailing edge *h* is the criterion determining friction losses and will be proportional to the square root of the parameter ZN/P.

Film Lubrication Conditions in Journal Bearings.

The conditions still hold good when both surfaces are curved instead of plane, as in journal bearings, provided a wedge-shaped fluid film is assured. If, as in Tower's experiment, the film thickness increases for a short distance inward from the inlet edge, the steepness of the pressure curve will be lessened in this region and the point of maximum pressure will be thrown nearer the outlet edge of the bearing surface. Thus the load W on the journal will produce hydrodynamic forces in the oil, as indicated in Fig. 4.

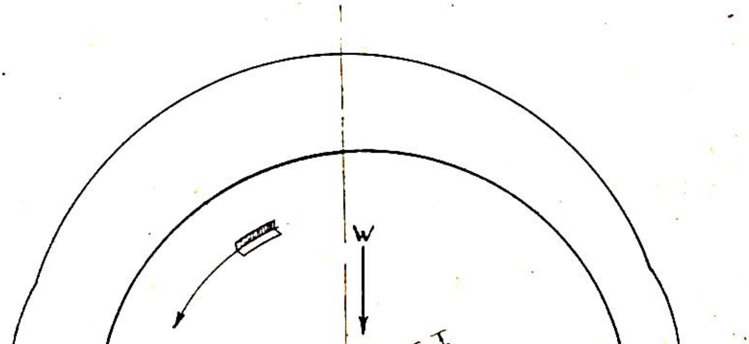
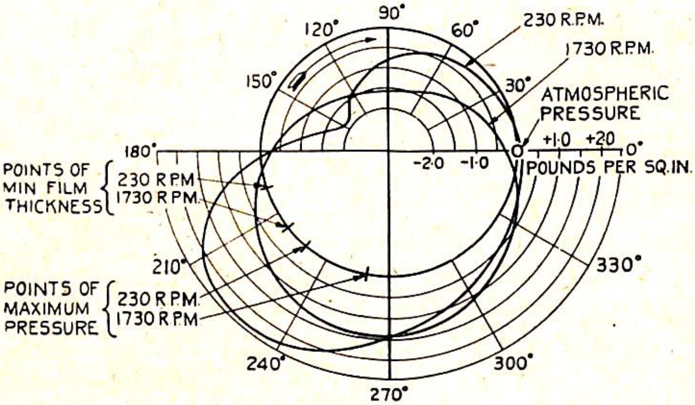


Fig. 4—Diagram showing shape of Oil Film Pressure and Conditions of Equilibrium for Journal Bearing.

The line of action of the resultant of these forces passes through a point which lies on the outlet side of the centre of arc of contact and, since the line must, of necessity, coincide with the vertical load W, the bearing arc must be arranged unsymmetrically with respect to the load. In other words, the conditions of equilibrium will be satisfied so long as the horizontal and vertical components of the fluid pressure equal and oppose the components of the force on the journal directed towards the bearing arc. The size and distribution of the forces will change as the journal climbs up on the bearing surface in the direction of rotation, and equilibrium is maintained while vertical resultants are obtainable from the fluid pressure. Hence, the loci of the centre of the rotating journal in relation to the bearing centre can be regarded as an equilibrium curve.

If the bearing arc is 180° or more, negative pressures will be found after the point where the film starts to diverge again, that is after the point of minimum film thickness. Gümbel, quoted by Stodola, however, prefers to neglect this as, normally the top half of the bearing is relieved and, hence, the pressure between points A and B will be atmospheric. In any case it will be very small, but may increase as the arc of contact increases.

Kingsbury experimented with a hollow cylinder and a plug which revolved in it, the lubricant being gaseous. This arrangement enabled him to take exact measurements, which showed the point of maximum pressure to be beyond the point of minimum film thickness (see Fig. 5). Incidentally, the substitution of air or other viscous fluids with Michell bearings has also proved successful.



FLUID PRESSURES
AT MID-LENGTH OF BEARING

Fig. 5—Kingsbury's Experiment.

For the conditions of minimum friction, the clearance, that is the difference in radius between journal and bearing, will have an optimum value and the eccentricity must necessarily be less than the difference in radii. The arrangement aimed at should make the film perfectly convergent. The clearance and, therefore, the minimum film thickness is limited in practice by the surface finish.

Swift gives a compromise of Goodman's and Cardullo's formula for the value of clearance as :—

$$d_1 = \frac{\sqrt{D_1}}{1000}\left(\frac{\sqrt{N}}{1 + 20}\right) \leq \frac{3}{2000}\sqrt{D_1} \qquad \text{(i)}$$

where N = 100 R.P.M. to 3000 R.P.M.
D_1 = bearing diameter, inches.
d_1 = diametral clearance, inches.

THRUST BEARINGS.

Description of Pads.

In practice, the actual pad area is about 80 to 85 per cent. of the gross annular area and the number of pads is generally 8 to 12. While determining the dimensions of the sector-shaped surfaces, it is necessary to take into account the ratio of length on the mean diameter and the radial length. The former is governed by the number of pads that can be fitted round the disc ; the radial dimensions depend on the diameter of the shaft and the space available in the pedestal ; it is limited by the maximum allowable surface velocity on the mean diameter.[1]* The best results were obtained when the radial width is the same as the length. It has been established that for any change in temperature the allowable unit pressure increases as the radial proportions increase, so that the larger the pads the better.

The rise in temperature in the lubricant is directly proportional to the square root of the load and is dependent on the viscosity and the surface speed. The film thickness of the trailing edge and, hence, the total clearance is limited, not so much by the reduction of frictional resistance, as by the total axial float permissible. The total oil clearance varies generally between 0·012 in. and 0·020 in., but may exceed the latter figure if other factors permit.[2] The minimum film thickness is dependent on the finish and workmanship of the white metal pad faces and the thrust collar. It must also be remembered that the physical cleanliness of the oil plays an important part. It has been noted that the white metal was scarred in the region of the trailing edge by minute particles in the lubricant. The edges of the pad will have to be rounded off carefully to allow no metallic rubbing at any time. With a clean oil supply the minimum film thickness may be as small as 0·0004 in.

[*] Where this note occurs, see References, page 35.

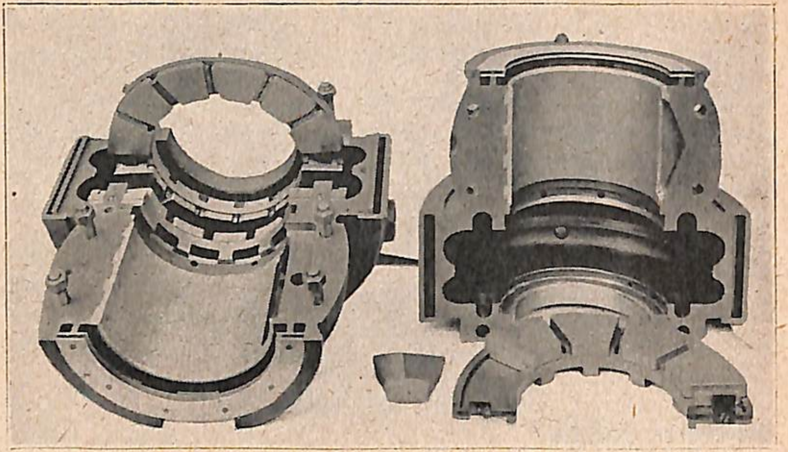


Fig. 6—Combined Thrust and Journal Bearing (Metropolitan Vickers Elec. Co.).

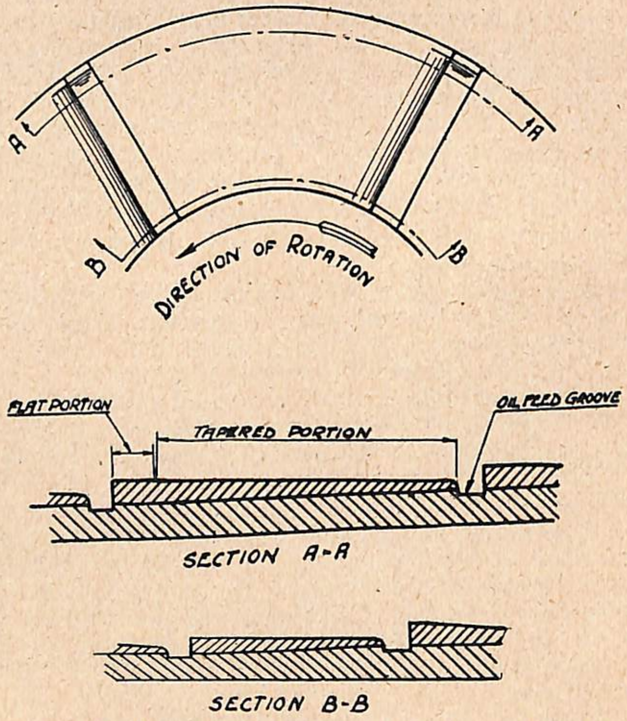


Fig. 6a—Principles of the Tapered-Land Thrust Bearing.

The position of the pivot axis is usually in the region of 2 : 3 on the mean diameter. The eccentricity is never greater than Length on Mean/6, though in marine practice centrally-pivoted bearing pads are used to avoid handing. It is evident that the self regulation of the pads goes a long way. One of the main aims of the designer is to ensure that the pads take up the necessary position of obliquity with ease.

The pivot axis can, therefore, be a radial ridge on the back of the pad or a hardened point on the radial bisector of the distributed pressure (see Fig. 6).

In a thrust bearing for a turbine, the thrust is of a steady nature and almost entirely uni-directional, hence the pads on the unloaded side are usually smaller than those on the loaded side. This ensures :

(*a*) a ready check on the handing of pads and thus makes for correct assembly ;

(*b*) a reduction in the frictional loss when compared to the case where the pads are equal ;

(*c*) that the total unit load, on the loaded side, is less than with equal pads.

Frictional Losses.

The friction losses are estimated on the basis of published curves or formulae, which generally reduce to the fundamental :—

$$H = \frac{AU}{J} f \qquad \text{(ii)}^3$$

where H = Power absorbed in B.Th.U./second.
A = Bearing surface in sq. in.
U = Peripheral velocity in ft. per second.
J = Mechanical equivalent of heat, 778 ft./lb. per B.Th.U.
f = the average shear resistance of the film in lb. per sq. in., which is related to

$$\sqrt{\frac{ZU}{PL}} \qquad \text{(iii)}$$

where L = Length on mean diameter.
Z = Viscosity of oil.
P = the pad pressure.

Fig. 7 shows a curve of frictional losses in a thrust bearing, calculated in line with the above equation. The validity of the parameter ZN/P is borne out by full-scale experiments, though the losses calculated according to various experiments differ.

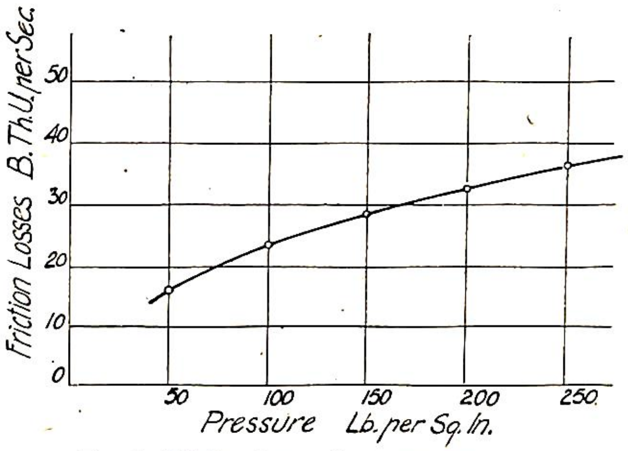


Fig. 7—Friction Losses in Thrust Bearings.

Lubrication.

In turbine work, as cooled oil under pressure is available, this is utilised. The quantity of oil to be admitted is computed so as to carry away the heat generated by friction, allowance being made for the heat received by radiation and convection. The maximum temperature rise allowed for, in practice, is about 25°F.[4] Here, it should be remembered, that the viscosity of the oil decreases with the rise in temperature. With the decrease in viscosity the load-bearing capacity of the oil film decreases. In the matter of choice of the lubricant, however, theory fails to give any guidance.

A separate oil supply is provided for, on both sides of the thrust collar, often on the inner peripheri of the pads and a continuous bath lubrication is ensured, while the oil is generally drained away on the top side of the housing. In fact, the amount of lubricant admitted is far in excess of the minimum needed for the formation and the maintenance of a pressure film, so that heat generated may be effectively removed. (Fig. 6).

When ascertaining the correct diameter of the oil inlet, the pumping effect of the thrust collar is taken into account. As the load on either side of the collar may frequently be unequal, a baffle plate is fitted in the housing, adjacent to the peripheri of the thrust collar. This arrangement enables one side to be supplied with more oil than the other.

Mechanical Properties of Pads.

In general, the pressure on the pads varies between 150 to 250 lb. per square in.[5], although higher pressures may be encountered.

Hence the mechanical strength of the bearing is important, especially in view of the fact that the thrust may increase under working conditions and the unit pressure thus exceed conditions for which it was designed.

The pads are manufactured of steel or brass and are made in form of a complete ring. After being white-metalled and accurately machined, the ring is cut into segments. The sliding surfaces are finished off by hand-scraping. The thickness of the white metal is usually about $\frac{1}{16}$-in. The part played by the white metal is that of an additional safeguard and, hence, its thickness will be such that, should it have to yield, no damage will be done to the turbine. The white metal used is a tin based alloy, the other components being antimony and copper.[6] Great care must be taken in preparing the white metal lining owing to the difference in rate of thermal expansion of the white metal and the pad body, and also the unequal rates of cooling. Further, if air gets between the white metal and the pad, this portion will be depressed when fluid pressure is applied and an uneven surface will result. The structural strength of the pads and the ring on which they are supported is of primary importance, as any distortions of them would disturb the formation of a pressure film.

Axial Setting.

Where the axial setting of the rotor is of importance, as in turbines, means for observing this and correcting for the effects of possible wear on the pad-faces are sometimes introduced. Excessive total load may cause interruptions of the pressure film, which may be the cause of such wear. One maker has introduced an adjustable thrust cage, whilst another has provided for mechanically-operated indicators showing the wear on the loaded faces.

The Hydraulic Tube Position Indicator.

The Metrovick hydraulic tube position indicator is sometimes fitted by the makers to their turbines (see Fig. 8). The principle of this patented device is based on the fact, that in practice wear, if any, occurs on the thrust pad faces and not on the collar. Oil is pumped at constant pressure through a nozzle, the open end of which registers with a small clearance against the face of the thrust collar on the pressure side. Should wear occur, the clearance would be reduced and the amount of oil escaping through the nozzle would be less. With the restriction of free passage for the oil, the pressure behind the nozzle will rise, as the original distance between the nozzle and the thrust collar is reduced. The pressure will rise until contact between the orifice and the thrust collar bars the flow. The oil pressure behind the nozzle will then be equal to that of the oil supplied. The rise in pressure behind the nozzle is recorded by a gauge and the arrangement is specially

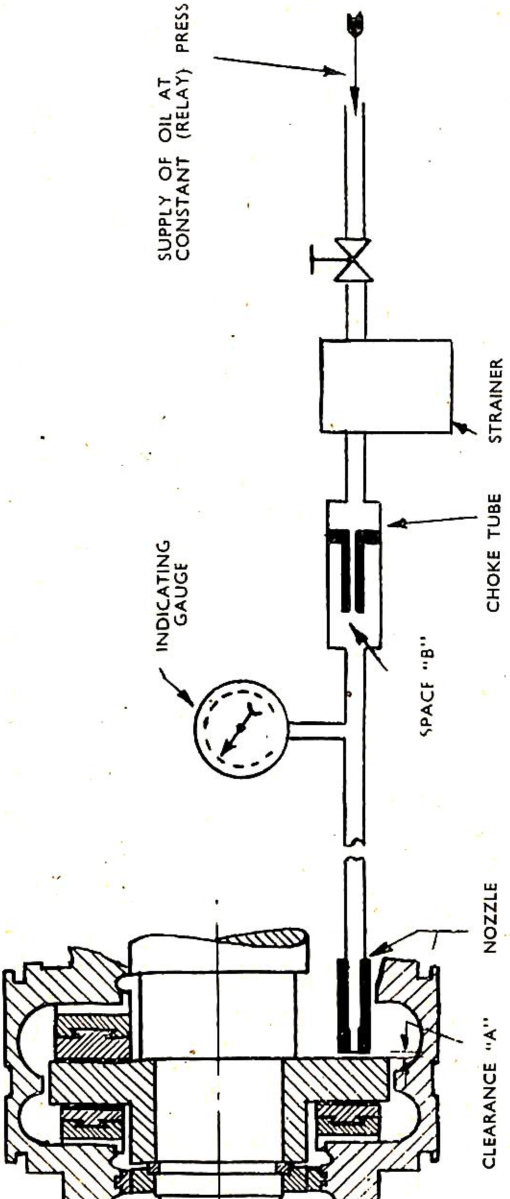


Fig. 8—Diagrammatic Arrangement of Metrovick Axial Position Indicator fitted to a Turbine Thrust Bearing.

designed to make the gauge readings independent of variations in the viscosity of the oil supplied to the nozzle. Guy reported that by using this device he showed that, contrary to general belief, the thrust bearing does not fail instantaneously, though, of course, this does not apply when the oil supply fails. Wear was taking place by minute surface flow from the white metal on the surface of one pad, which was deposited on the leading edge of the following pad. This building up of the leading edge, which finally caused the interruption of the oil film, may go on for weeks.[7]

It is necessary that the bearing surface always remain truly perpendicular to the axis of the spindle. To achieve this without any special adjusting device, the thrust bearing is combined with a journal bearing which is supported on a spherical seating (Fig. 6). According to Soderberg,[8] in this combination, the journal-bearing carries a comparatively small load, hence there is no inconsistency in the fact that the limit of peripheral speeds for the thrust bearing is lower than for the journal bearing.

The Tapered-Land Thrust Bearing.

The merit of this type of thrust bearing is the simplicity of its construction. The thrust collar rotates between two rigid plates. The plate on the thrust side holds a split steel ring which has a babbitted face divided into lands by radial oil-feed grooves. The surface of each land is tapered so that it slopes relatively to and towards the rotating collar, both in the direction of rotation and from the inner to the outer radii at the leading edge of the land. A suitable approach curvature from the oil feed groove to the lands is provided at their leading edge (see Fig. 6a). On reversing marine turbines, the flat portion of the land may occupy as much as one-half of the total land area and suitable approach curvature from the oil feed grooves is provided at both ends of the lands. In machines where the thrust is uni-directional, the plate on the unloaded side holds a babbitted plate with flat lands. The diameter of this is smaller and it serves as a check on end-play. In marine practice the direction of thrust load may change with reversal so both rings have tapered lands with a relatively large proportion of the land parallel with the faces of the thrust collar.

The tapered land thrust bearing is largely used by the General Electric Co. of Schenectady. The design of the correct taper has been developed from Gümbel's theory, supplemented by results of tests. The field of application ranges for speeds from 1,500 R.P.M. to 20,000 R.P.M., with thrust loadings from 150 to 450 lb. per sq. in. on turbines varying from 100 kW. to 160,000 kW. rating.[9]

FILM LUBRICATION CONDITIONS IN JOURNAL BEARINGS.

Whereas film lubrication has been successfully applied to thrust bearings, its application to the journal bearing was less straightforward.

Michell Type Journal Bearing.

Although the principle of the wedge-shaped oil film as the load-bearing medium still holds, difficulties are encountered which arise mainly from the necessity of using a fixed and curved bearing surface. Michell has overcome these difficulties by running the journal in a ring of pads carried by a steel ring.[10] The bearing faces of these pads are bedded to the journal and the faces adjacent to the steel ring are curved so that line contact only occurs. Sufficient clearance is allowed to enable the pads to rock or tilt slightly and thus operate in much the same way as the flat-bearing pads of the thrust bearing (Fig. 9). Tilting pad bearings, says Michell, present analogies with ball and roller bearings, particularly in their adaptability to accurate machine manufacture without the particular purpose or destination of any bearing being known in advance. The journal load per unit of axial length is given by :—

$$B = C \,.\, \mu \,.\, V .\, r^{2} \; e/n \; (d^{2} - e^{2})^{3/2} \qquad (\mathrm{iv})$$ [11]

where μ is the viscosity, in centipoises,
V is the peripheral speed, in cm. per sec.,
r is the radius of the journal, in cm.,
e is the eccentricity of the journal, in cm.,
n is the number of pads (which, in the case of the above equation, is taken as always greater than 8),
C is a constant which has a value of 14·1,

if the pads are pivoted behind their centres in the normal manner and their axial dimensions are large compared with their circumferential lengths. This value is given for the C.G.S. system.

Journal Bearings.

While the detailed description of the special types of journal bearings cannot be attempted here, it is intended to examine in greater detail the operation of bearings with clearance brasses, *i.e.*, of bearings which are finished to a slightly larger diameter than the journal.

From the previous general discussion it can be seen that the film after reaching the point of minimum thickness could be divergent towards the outlet end. An excessive length of divergent film may actually cause the film pressure to fall below the pressure conditions existing at the edges of the brass. Thus the magnitude of the bearing arc is an important factor and is determined by reliefs

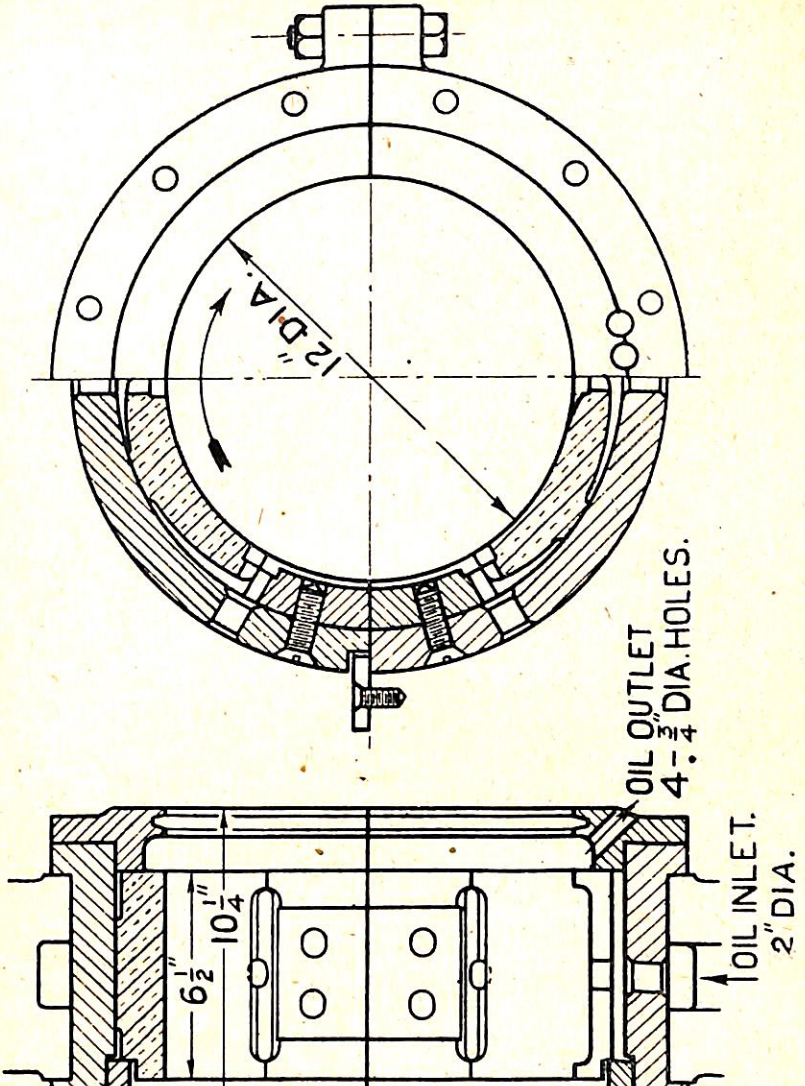


Fig. 9—Michell Type Journal Bearing.

at the sides. These reliefs are not carried the full length of the bearing but have small axial grooves at each end, so as to increase the flow and carry away heat. The necessity for avoiding sub-atmospheric pressures in the oil film arises from the danger of air being drawn in round the edges which disturb the continuity of the oil film. The outline of the bearing, shown on Fig. 10, represents an ordinary journal bearing. The clearance of ·002-inch per inch of diameter is generally adopted when the pressure is up to 150 lb. per sq. in. of projected area, and the minimum surface speed is not less than 20 feet per second.[12] After accurate machining of the bore, no further scraping of the bearing surface is found necessary. The upper limit of velocity is not known, but a practical limit will be set by frictional losses, which is well below the limit of film failure. To limit the surface speed, the radius of the journal will be kept as low as possible, and there is advantage in doing this from the frictional point of view. In present practice, the surface speed may be up to 177 feet per second.[13]

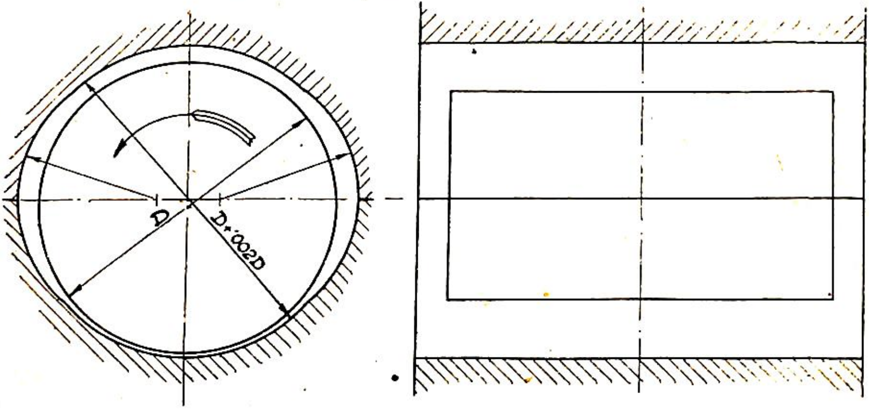


Fig. 10—Diagram showing Clearances and Reliefs in Simple Journal Bearing.

Bearing Capacities.

Although the estimation of bearing capacities centres theoretically on the safe film thickness, in practice the following points have to be taken into consideration :—

- (i.) The degree of finish on the combined journal and bearing surfaces and the machining. Stodola recommends a surface finish to about ·0004-inch to ·004-inch.[14]
- (ii.) Variations of diameter along the shaft or the bearing. Expansion of the journal owing to axial heat transmission. Possible local distortion owing to static pressure.
- (iii.) The deflection of the shaft. Direction and constancy of loading. Rigidity of the brass and bearing housing.
- (iv.) The properties of the lubricant.

Oil Inlet.

The correct position of the oil inlet is important, for, failing an adequate oil supply for the formation of the film, the thickness is reduced, which will result in increased friction. The oil inlet is situated either just below, or on the horizontal joint on the rising side, and the top half of the bearing is relieved to form an annular space so as to allow for free circulation of oil. Only a small proportion of the lubricant goes from here to form the oil film which carries the load ; the rest is for cooling. (Fig. 11).

Conditions of Operation.

The lubricant, in its passage through the clearance space, is subjected to shear. The torque required to drive the journal is proportional to the resultant shear stress on the film. The work done by the journal is transformed into heat which is transferred to the oil film. It is therefore necessary to provide for cooling the lubricant during its passage from outlet to the inlet side of the bearing. Since bearings for turbines are intended for a long life, they operate with a comparatively thick wedge-shaped oil film. This may tend to increase the friction due to shear in the layers of the lubricant. In general, on the score of reliability, a large film thickness and not a minimum coefficient of friction will be the deciding factor[15].

The coefficient of friction $\mu = \dfrac{\text{Shear stress}}{\text{Total load}}$

and $\mu = K\ (ZN/P)^n$ (v.)[16]

where, according to Boswall, n is 0·58 approximately ; the values for K are given for central loading as :—

Bearing arc.	K
45°	0·0028
90°	0·0028

End Leakage.

The hydrostatic pressure at the ends of the bearing, where they open into the housing, is nil. The resulting pressure gradient causes the oil to flow axially, its path being of a helical shape. Only a small part of the leakage at the ends may be drawn back if negative pressure exists in the divergent region of the film. The rest of the end leakage will have to be accounted for in the amount of lubricant admitted. The extent of leakage is in direct proportion to the velocity of the journal, the radius of the journal and the radial clearance ; it is also dependent on the ratio of length to diameter of the bearing arc and the eccentricity.

Thus, according to Karelitz, for a wide range of loading conditions, with an arc of 120°, the eccentricity ratio between 0·4 and

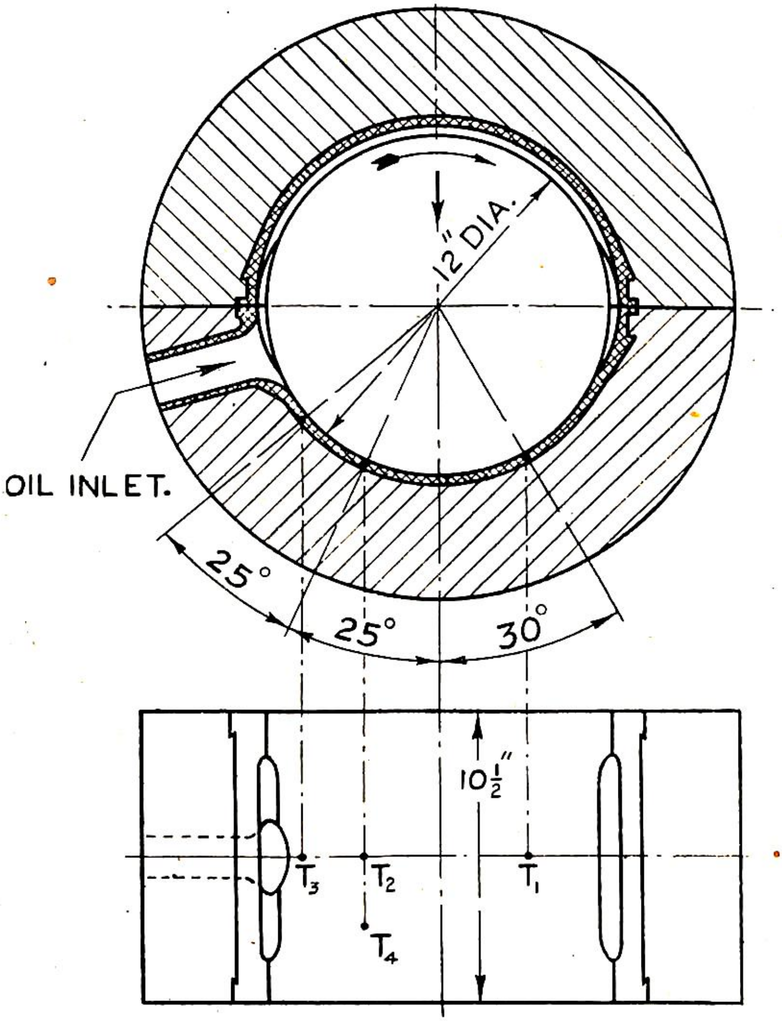


Fig. 11—Journal Bearing.

0·8, and the ratio of length to diameter between 0·75 and 1·5, the end leakage may be estimated by the simple formula

$$Q = 0{\cdot}27\ U\ R\ r \qquad \text{(vi.)}^{17}$$

where

Q is the side leakage in cubic inches per sec.
U is the linear velocity of the journal in inches per sec.
R is the radius of the journal in inches.
r is the radial clearance in inches.

Eccentricity.

The eccentricity must necessarily be less than the clearance or radial difference, hence the minimum film thickness h may be expressed as :—

$$h = \tfrac{1}{2}\ (D_B - d_J)\ -\ e \qquad \text{(vii.)}$$

where

D_B is the diameter of the brass in inches.
d_J is the diameter of the journal in inches.
e is the eccentricity of the centre of the journal in inches;

if

$r = \frac{1}{2}\ (D_B - d_J)$ the radial clearance,

and

e/r the eccentricity ratio,
then

$$h = r\left(1 - \frac{e}{r}\right) \qquad \text{(viii.)}$$

The eccentricity is directly proportional to the radius of the journal and will depend upon the value of the product ZN/P, the ratio of length to diameter, the clearance ratio (r/R), the angle subtended by the brass and the angular position of the line of action of the load from inlet edge of brass. The various theories appear to agree in suggesting that there is some eccentricity at which any clearance bearing should work in order to give optimum conditions of film thickness and friction, but there is no serious change in the conditions over a considerable range of eccentricities in the region of this optimum.[18]

Clearance.

The bearing clearance, that is, the amount by which the journal diameter is smaller than the diameter of the bearing arc, varies rather widely, according to the type of bearing and its application. In turbine practice it ranges, according to manufacturer, between ·001″ to ·003″ per inch of journal diam.[19]. According to Boswall, the minimum friction will occur when the clearance, r, has a definite value. He recommends a suitable clearance from this point of view of about ·0015 times the radius of the journal.[20]. For an

180° centrally-loaded clearance bearing, if designed for an eccentricity ratio of ·40 (the optimum theoretical), r is given by Swift as $1{\cdot}87\ R\sqrt{ZU/P}$.[21]

As the clearance is often of the same order as tolerances in manufacture, some variation of the clearance value from that given in the design may be expected. However, the performance is only likely to be affected if the error exceeds about 30 per cent.[22]

Bearing Arc.

In general, the bearing arc should be less than 120°. Bearings of the type shown on Fig. 11 have an arc of between 60° to 100°. The aim is to achieve a perfectly convergent pressure film over the bearing arc, and thus decrease the frictional resistance. It must not be forgotten that for a given length of brass and value ZN/P a decrease in the arc of embrace causes an increase in the maximum pressure. This, and the ensuing decrease in the minimum film thickness, may result in excessive temperature rise of the lubricant. The reduction of bearing arc should normally only be undertaken when the load capacity is ample, and the available quantity of free lubricant is sufficient to cope with the increased temperature.

Operating Conditions.

The relative position of the journal and the bearing for any given working condition is dependent upon the loading and the clearance. Equilibrium conditions will be reached as soon as the line of centres has swung round in the direction of rotation sufficiently to allow for the formation of the convergent film. In machinery which is subjected to frequent starting and shut-down provision must be made for a good supply of lubricant for the period during which greasy or boundary lubrication conditions occur. In turbine practice, an auxiliary oil pump is provided for the initial flooding of the bearings, though film conditions are established soon after starting. On some large turbines, barring gear is used and oil from the jacking pump is supplied through holes in shallow recesses at the lowest point of the bearing arc. (Fig. 12).

Lubrication.

The secondary duty of the lubricant is the disposal of heat generated by friction. The energy dissipation and temperature rise of a bearing depends more on the speed of rotation than on the load or the oil viscosity. In turbine work, taking advantage of the force feed lubricating system, the temperature rise is controlled by increasing the oil flow. The limiting point will be reached when the viscosity of the cooled inlet oil needlessly increases friction. According to Lasche, the inlet oil temperature should not be below 95°F. The inlet oil pressure is in the region of 5 to 10 pounds per square inch.

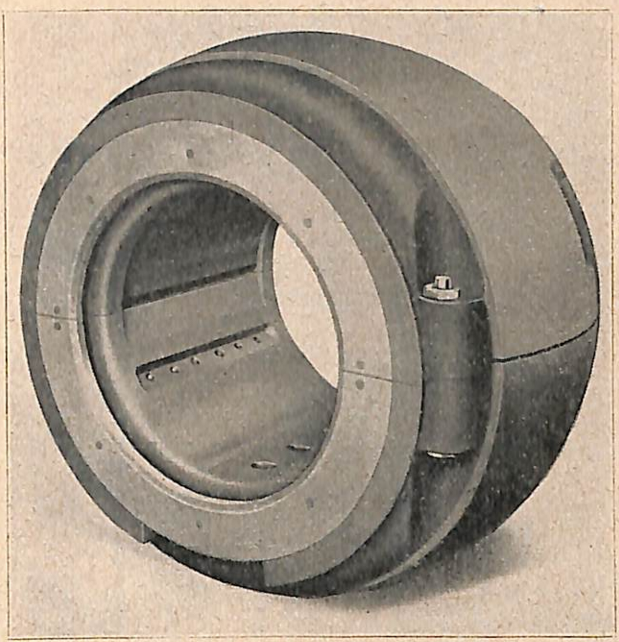


Fig. 12—Typical Large Main Turbine Bearing, showing Oil Inlet, Scraper Groove and Inlets for Jacking Oil.
(Metropolitan Vickers Electrical Co.).

The circulation of the lubricant in a correctly-designed bearing is entirely due to the rotating journal. The introduction of oil grooves in the bearing arc portion of the brass would greatly interfere with the formation of a pressure film.

To encourage the free circulation of oil in the system, and thus help the extraction of heat through the lubricant, a special drain passage is arranged in certain bearings. (Fig. 13). The function of this groove is to draw off the oil once it has been round the journal. The same idea has prompted the application of the moveable scraper, which scoops the oil off the journal and directs it into the outlet channel. (Fig. 14). In either case, it should be noted that the position of the oil outlet must be chosen at a point of neutral pressure, lest the rotating journal develops a pumping action and starves the bearing of lubricant. Special circumferential grooves may sometimes be provided in the top half of the bearing to give better distribution to the incoming oil and to flood the journal. These grooves are sealed against air ingress to prevent aeration of the oil, and thus stop the cause of oxidisation in, and acidity of, the lubricant.

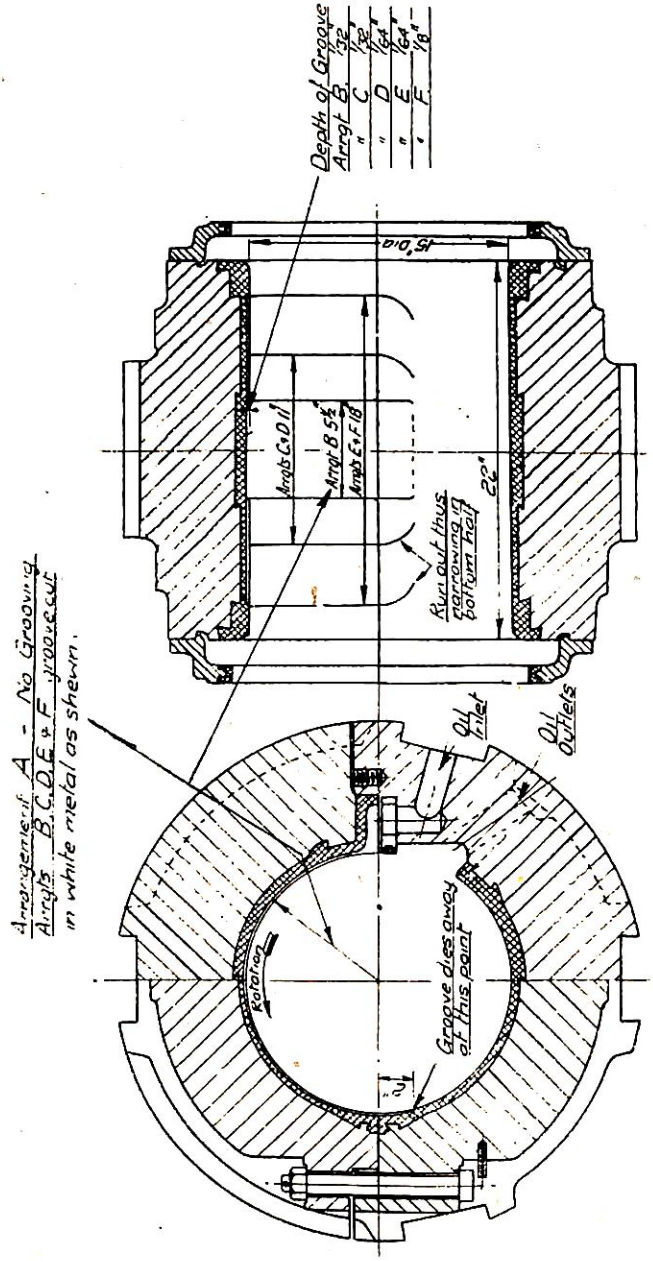

Depth of Groove
Arrgt B. 1/32"
" C 1/32"
" D 1/64"
" E 1/64"
" F 1/8"
15" dia
Arrgts C+D 11"
Arrgt B 5½"
Arrgts E+F 18"
22"
Run out thus narrowing in bottom half
Arrangement A - No Grooving
Arrgts B,C,D,E+F groove cut in white metal as shewn.
Oil Inlet
Oil Outlets
Rotation
Groove dies away at this point
3"

Fig. 13—15″ × 22″ Scraper Type Journal Bearing, with Oil Distribution Grooves.

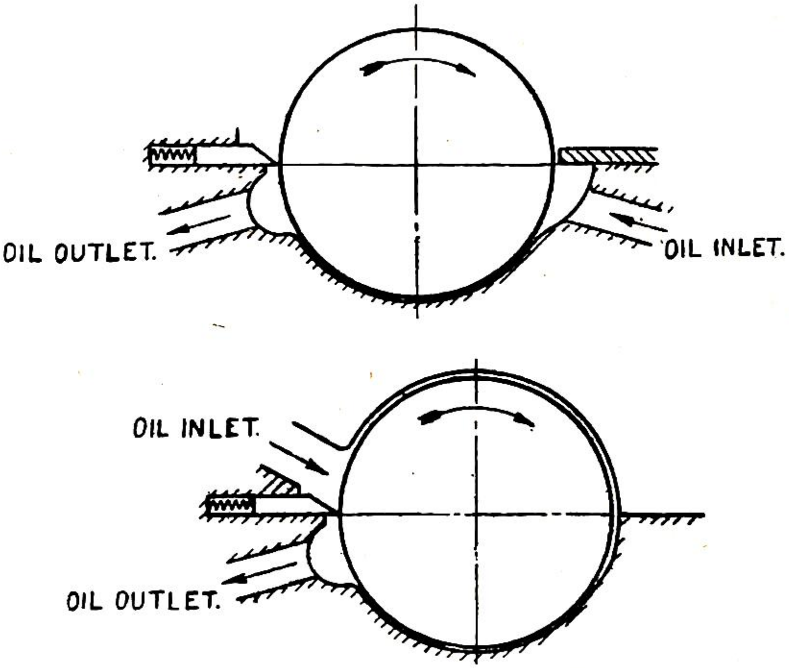


Fig. 14—Principles of the Moveable Scraper.

Ring Lubrication.

In bearings where the velocity of the jornal is in the order of 350 to 1000 R.P.M., and the loading in the region of 50 to 500 lb. per sq. inch of projected area,[23] rings have been designed to supply a sufficiency of oil for the maintenance of an oil film.

The transfer of oil from ring to journal depends mainly on the adhesion and cohesion of the oil, the actual mechanism of the transfer is as yet imperfectly understood. Centrifugal forces hinder the transfer by preventing gravitational flow from the sides of the ring and at higher ring speeds by overcoming adhesion. The main source of oil is that carried on the inner surface of the ring. At low journal speeds the side areas of the ring contribute substantially to the amount of oil transferred, whereas at higher speeds this advantage is lessened and the transmission rate is roughly proportional to the width of the ring. Grooving the inner surface of the ring assures against excessive oil leakage at the sides of the ring, and so comparatively thick oil films will be transferred on to the journal. At low journal speeds the amount of oil supplied by this method is far in excess of the amount lost due to end leakage.

At higher journal speeds, however, where the end leakage is greater, and provision ought to be made for excess oil to carry away heat, the design of oil rings is a vital factor.[24]

Oil Quantity.

The correct oil quantity is assessed on the basis of frictional losses and the permissible temperature of the bearing. .

When calculating frictional losses, the various formulae again reduce to the basic

$$H = \frac{A\,U}{J} f \text{ B.Th.U./sec.} \qquad \text{(ii.) ante.}$$

with various empirical constants introduced by the individual designers. (Fig. 15).

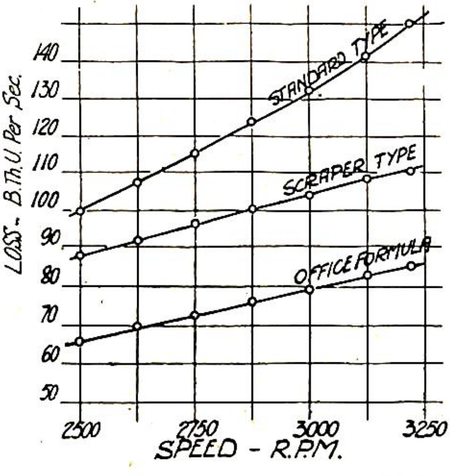


Fig. 15—Friction Losses in Journal Bearings. (Based on Curves published by F. J. Cowlin).

Boswall recommends the following formula for estimating the correct quantity of lubricant :—

$$Q = C\,U\,R\,B\,(ZN/P)^m \text{ gallons per minute.} \qquad \text{(ix.)}^{25}$$

where U is the surface speed of the journal ft. per sec.
R is the radius of the journal in inches.
B is the axial width of the brass in inches.
Z is the viscosity in poises.
N is journal revolutions in R.P.M.
P is the load per unit area, *i.e.*, the total load divided by product RB.

Values of the constants C and m are given in Table 4, page 40.

Two approximate formulae may be quoted here for the calculation of the amount of power lost in film lubricated bearings :—

$$\text{KW Loss} = \frac{0{\cdot}38\ A\,P^{0.4}\,V^{1.2}}{10^6} \qquad (x.)^{26}$$

and

$$\text{KW Loss} = 5{\cdot}62\ 10^{-6}\ L\,D^2\,N \qquad (xi.)^{27}$$

where A is the projected area in square inches.
P is the bearing pressure in lb./sq. inch of projected area.
V is rubbing speed in feet per min.
L is the length of the bearing in inches.
D is the bearing diameter in inches.
N is the speed of the journal in R.P.M.

Bearing Temperature.

The permissible temperature of the oil leaving the bearing varies according to the designer between 140°F. and 160°F.[28] It is the practice of the General Electric Co., Schenectady, to limit the temperature rise in bearings to 104°F., and on large machines to 86°F.[29] Boswall evolved the following relationship :—

$$t^0 = \frac{60\ KS}{C}\ P\ (ZN/P)^{(n-m)} \qquad (xii.)^{30}$$

where t^0 is in degrees C.
S is the specific heat of the lubricant in energy units per gallon per degree C.

Other values are as in previous equations, and the values for the various empirical constants are given in Table 4.

Thus it can be seen that the rise in temperature depends also on the parameter ZN/P. A bearing operating under stable conditions is one in which the rate of heat generated will correspond to the rate of heat dissipated. The temperature rise will also be affected by heat transmitted through the shaft and by heat received through radiation and convection from adjoining regions of high temperature, and these should be taken into consideration.

Dimensions of Bearing.

The tendency in modern high-speed bearing design is to reduce the value of the ratio

$$\frac{\text{Length of bearing}}{\text{Bearing diameter}}$$

to values between 1 to 1·5, the minimum being about ·75[31]. The reduction of the axial length beyond certain limit would result in alteration to other factors of operation such as the reduction of film thickness and the increase in friction.

BEARING METAL.

The bearing metal should satisfy the following requirements :—

(1) It should be easily melted, poured and machined.
(2) It should be capable of good adhesion to the shell.
(3) It should have high compressive strength at working temperatures, good wearing qualities, and a low coefficient of friction.
(4) It should have sufficient plasticity to compensate for inequalities of alignment.

The composition of the white metal is tin, antimony, copper, and sometimes lead. The composition varies within certain limits. Thus, an arbitrary increase of the copper and antimony, which form the hard compound of the alloy, may create a state where the hard crystals are touching, and too little plastic matrix is left to enable it to adapt itself to a new shape. Under severe conditions these crystals will slide on each other, and failure by flaking or cracking will occur. The addition of copper in the tin-antimony alloys is necessary in order to prevent the segregation of the brittle tin-antimony compound.

The adhesion of white metal to cast iron is approximately one-third of the total strength in tension of the alloy, while, with a smooth-machined steel surface, the joint is more than three-quarters of the strength.[32] Adhesion to the shell may in some cases, and with certain materials, be unreliable. For this reason, and as an additional safeguard, the white metal is sometimes fixed in the bearing by dovetail grooves. These are arranged circumferentially, with an axial one at the top and bottom of the bearing, or on the horizontal joint.

White Metalling Practice.

In modern white metalling practice, four main methods of lining bearing shells are adopted :—

(*a*) Continuous white metal lining of steel strip.
(*b*) Pressure die-casting.
(*c*) Rotary lining.
(*d*) Hand-pouring.

(*a*) This method is employed in the manufacture of the modern thin-walled bi-metal bearings. The details of manufacturing technique are not available for publication, but the product is a steel-backed bearing having a total wall thickness, including white metal, of the order of ·050″ to ·070″, the white metal thickness ranging from ·005″ to ·020″. The bearings so produced are known as pre-finished bearings, as they are finished to an accuracy in wall thickness within ·00025″ and with a finished back which fits perfectly into the bearing housing. No hand-scraping or other fitting is necessary with such bearings, which are almost universally used in the modern motor car engine.

(*b*) In this method, the bearing shell, which may be of steel, bronze, or brass, is placed, after suitable treatment, in a die which is then conveyed to a pressure die-casting machine. White metal is fed under pressure into this die, which is then removed, opened, and the lined shell ejected. This method of manufacture is suitable for bearings thicker than those which can be lined in the form of continuous strip, and is applicable to bearings of practically all sizes, except perhaps the very largest, such as those employed in rolling mills, etc. It may be noted, however, that bearings up to 15″ diameter and 15″ long have been lined by this method, taking 56 lbs. of metal as cast, the limiting factor being the quantity of metal which can be pumped into the die from a given die-casting machine. For very large bearings, two machines are frequently used in tandem.

(*c*) This method is particularly applicable to bushes, but is also used for the lining of half bearings. Again, after suitable preparation, the bush or half bearings, the latter usually in pairs, are placed in the head of the rotary-lining machine, which is then rotated at speeds of up to 1500 R.P.M. White metal is poured into the bushes while rotating, to give the required thickness as cast. Spinning is continued until the white metal is solid, and the bush is then removed and the next one inserted.

(*d*) This is the oldest method of manufacture, which has been adopted for very many years. It is used for large bearings and for special bearings to which the foregoing methods are not applicable.

Preparation of the Shell.

Whatever the method of manufacture adopted, certain general principles must be observed in the preparation of the bearing shell :

(*a*) DEGREASING.—The shell, whether it be of steel, bronze or brass, must be thoroughly degreased prior to white metal lining. Degreasing may be carried out in the solvent type of degreaser, such as trichlorethylene, or in the alkaline bath type of degreaser, such as the modern trisodium phosphate or silicate aqueous solution. After degreasing, the bearing is thoroughly rinsed in clean water.

(*b*) ETCHING.—The shells are now etched, in the case of steel, in hot diluted sulphuric acid solution or cold hydrochloric acid solution ; or, in the case of gunmetal, bronze or brass shells, in nitric acid or nitrosulphuric acid, cold solution. The object of this etching is not, as is frequently supposed, to obtain a pitted surface and therefore mechanical keying, but to produce a chemically clean surface for tinning. The etching is continued just long enough to produce such a surface, and the bearing shell is then thoroughly washed in cold and then hot water.

(*c*) FLUXING.—The bearing shell is now treated on the back, or on those surfaces which have not to be tinned, with whitewash or other material which will prevent tinning. The surfaces which are to be tinned are now fluxed with a solution of zinc chloride, frequently prepared by dissolving zinc in concentrated hydrochloric acid until no more zinc will dissolve, this solution being commonly known as "killed spirit." It is advantageous to have some slight residual acidity in the resulting solution. The surfaces are swabbed with this solution and the shell is immersed in the tinning bath.

(*d*) TINNING.—In the case of steel shells, pure tin is used and a tinning temperature of about 300°C. is suitable for most jobs. The fluxed shell is immersed completely in the tinning bath for just long enough to permit the shell to attain the temperature of the molten tin. There is no advantage in leaving the shell to soak for a longer period than necessary. The shell is then withdrawn, thoroughly shaken to remove surplus tin, and in the case of die-casting, the shell is inserted without delay in the die ; in case of rotary lining, it is put into the rotary head ; while when hand-pouring is used, the shell is assembled in the jig prepared for the purpose.

Bronze or brass shells are usually tinned in a tin/lead solder, the normal proportion being 50 per cent. tin and 50 per cent. lead. This permits working at a slightly lower temperature and also cuts down the rate at which the metal of the shell is being attacked by the tin. The latter point requires attention, firstly owing to the detrimental effect upon the bearing shell, and secondly, on account of the serious contamination of the tin bath. Again, the tinned bronze or brass shell is assembled in the die, rotary lining fixture or jig, with the minimum of delay.

Methods of Casting.

(1) DIE-CASTING.—The temperature of the metal depends upon the particular composition which is being cast and may vary from 350°C. to 500°C. Also, with a given composition of metal, the optimum temperature will depend upon the size of the bearing and the thickness of the lining as cast. The temperature of the die and of the mandrel must be suitable for each particular job. The tinned bearing shell is transferred rapidly from the tin bath, after shaking, to the die, which is clamped on to the die-casting machine, and the white metal is forced into the die under pressure by the displacement of a plunger in a cylinder.

The white metal is forced up through a gooseneck into the die, which it fills completely. The type of feed into the die again depends upon the particular bearings which are being lined, and is a matter of detail die design. After casting, the white metal is left to solidify and the die is removed from the casting machine. The bearing shell is ejected and permitted to cool naturally. It is important that the shell should not be distorted or be subjected to shock while still hot.

(2) ROTARY LINING.—The shell, after degreasing and tinning, is immediately inserted into the head of the rotary lining machine, while still hot. This head may be so arranged that by pressure on a foot pedal the back plate moves back and permits of rapid insertion of the bush; whilst on releasing the pedal the bush is held between the back plate and a front plate which has a hole in its centre. As soon as the bush is in position the machine is set to rotate. White metal is poured through a hole in the front plate into the bush. The centrifugal force throws the molten metal against the inner wall of the rotating bearing shell. It is claimed for this method that it avoids the formation of air pockets, while the impurities, which are lighter than metal, collect on the inner surface and thence are removed by boring. The speed of rotation depends upon the bearing diameter and upon the composition of the white metal. Since it is necessary that the speed should be sufficient to cause adequate centrifuging, but at the same time should not cause excessive segregation of the lighter constituents of the alloy, its exact determination is essential. This can only be done by experiment on individual jobs.

(3) HAND-POURING.—Little comment is necessary on this method. Preparation of the shells is similar to that stated above. The question of whether the shell is lined while lying on its back or standing on its end is again only determined by the particular job under consideration, and the white metal is transferred to the assembled tinned shell by ladle from the white metal pot. It is common practice to provide a small ring or fixture in order to ensure that adequate molten metal is available to feed the white metal during cooling and solidification. Also, it is usual to puddle the head of the liquid metal during the preliminary solidification period for the same reason. It is an added advantage to provide directional cooling from the bottom upwards by application of a water spray or wet rags, commencing at the base of the outside of the shell or bush.

Thickness of White Metal.

The thickness of white metal in the finished lined bearing is determined by the duties which the bearing will be called upon to perform. It has already been mentioned that in the thin-walled, pre-finished automobile bearings white metal thicknesses as low as ·005″ have been adopted, and in America a micro-bearing has been used, where the white metal thickness was ·002″. Such ultra thin linings are only applicable, however, to small thin-walled bearings. It should be emphasized here also, that such thin linings are subject to heavy wear if there are any impurities in the oil.

The following are examples of white metal thicknesses adopted in this country :—

> In automobile bearings, the thickness is in the order of ·015″ to ·020″ ; in small high-speed Diesel engine bearings from about ·020″ to ·040″ are common ; in larger bearings the trend is towards linings of the order of ·060″ to ·080″. This is recent practice in place of the old linings which were anything from $\frac{1}{8}$″ to $\frac{1}{4}$″ thick.

The thinner lining has the advantage over the thick white metal lining, in that it confers higher fatigue resistance upon the bearing metal, in addition to the saving in tin. As a guide to current practice, see Table 5.

Bearing Shell.

In large machines, such as turbines, the entire bearing shell may be of steel and may be supported on a spherical seating to enable self-adjustment (Fig. 12). Bearings may also rest on spherical or cylindrical pads behind which liners can be inserted when the alignment is checked. It is an advantage to provide for adequate lubrication of the spherical seating in order to insure for compensation for deflection in the spindle.

The scraper rings shown, which are fitted in conjunction with main bearings, are intended to reduce leakage of oil along the shaft (Fig. 13).

PROPERTIES OF THE LUBRICANT.

The description of the properties of the lubricant is beyond the scope of this pamphlet ; however a few general facts which are of consequence to successful operation may be stated.

The same lubricant may produce different coefficients of friction when separating different metals, and this is sometimes explained by what is called the boundary film effect. By this is meant, that lubrication is to some extent a chemical phenomenon, that is, the lubricant interacts chemically with the bearing metal and the journal to produce a film of great stability on the surface. This surface action is attributed to what is termed the "oiliness" of the lubricant. W. H. Herschel[33] gives the following definition of oiliness :—

> "If two oils, having the same viscosity at the running temperature of the film, give different coefficients of friction when tested under identical conditions, the one giving the lower coefficient is said to be the more oily."

According to Bradford and Vandegrift,[34] however, the difference in the coefficient of friction is due to the pressure-viscosity effect and not to this property of oiliness. As already stated, in the perfectly convergent oil film, the pressure varies from nil at the leading edge, through a maximum, back to nil at the trailing edge. This pressure is dependent upon the viscosity Z of the oil within the film ; but Z is in turn affected by the variation in the pressure. Hence, other conditions being equal, the higher the viscosity, the higher the pressure. As, however, in a comparatively thick film, Z is affected by the temperature rise due to friction, this state is never maintained.

It would appear that the ideal lubricant for supporting high loads at high speeds would be the one whose viscosity does not decrease much with the rise in temperature, but increases with the rise in pressure. Present-day lubricants cannot fulfil these requirements ; in fact, the very opposite occurs.

Oil Whip.

Under all normal conditions, bearings are inherently stable. Certain special combinations of circumstances, happily rare, can produce instability, a phenomenon not thoroughly understood. Should the journal become temporarily unbalanced, owing to some external cause, the film in the bearing will tend to magnify the vibratory motion ; this is sometimes called oil whip. It seems possible from Bowden and Tabor's experiments on "The Contact of Colliding Surfaces,"[35] that no metallic contact need occur. Thus, if bumping occurs between surfaces which are lubricated with a film of mineral oil, enormous forces may be transmitted through the oil film. Even though contact may be prevented, the forces transmitted through the oil film may cause distortion to the metal surfaces. The viscosity of the lubricant is the important factor if vibration is present. If the viscosity is sufficiently high, it may prevent the surfaces from touching. Therefore, from this point of view, it is advantageous to choose a lubricant whose viscosity increases with rise in pressure and the rate of flow.

Example in the Choice of Lubricant with reference to Steam Turbines.

It is of first importance that steam turbine lubrication oil should retain the properties in service required by continuous circulation for long periods, possibly in the presence of moisture and other impurities and in contact with bearing surfaces where the conducted and radiated heat all contribute to the possibility of breakdown in the oil.

Knowing the duty involved, a Committee of Government Departments, Technical Institutions, Turbine Builders, Oil Companies and Turbine Users drew up B.S.I. Specification No. 489/1933, in which the essential characteristics necessary in oils to meet the conditions of straight-coupled and geared turbines are described. Of these characteristics the most important are viscosity, demulsibility, ability to resist oxidation and flash point. The light grade oil is often employed in direct connected machines up to 5000 kW. For larger machines of this type, the medium grade may be used, while geared turbines use the heavy grade. Ring-oiled bearings are catered for by the extra heavy grade. Where temperature conditions exceed the normal, the next heavier grade is generally used.

Moist air entrained in the circulating system, combined with small quantities of iron oxide and water, may, in unsuitable oil, produce a sludge, with the result that the oil increases in viscosity and acidity grows rapidly. If the oil used has a poor demulsification value, water will not separate freely. If iron oxide and other impurities continue to contaminate it, heavy sludging may occur, and its stability in service is seriously impaired. The life of the turbine oil charge may be seriously affected if proper care is not taken to prevent the action of these impurities upon it.

The rate of circulation of oil through the complete lubricating system must be arranged so that any impurities, entrained gases, water, etc., can be eliminated before the oil re-enters the bearings.

Acknowledgments.

In conclusion, thanks are due to the Metropolitan Vickers Electrical Company for the use of their publications and illustrations; also to the Glacier Metal Company for their contributions on the subject of White Metalling Practice, and to Messrs. Fletcher Miller Ltd. for their suggestions on the choice of lubricant. Acknowledgments are due for the help of Mr. J. Polding, who assisted in the preparation of some of the illustrations.

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APPENDIX 1. Taken from **TABLE 1.**—Tilting : Pad Thrust Bearings for Steam Turbine Machinery, by R. Dowson. General Discussion on Lubrication and Lubricants. I.Mech.E. 1937. *Bearings of the Michell Bearing Coy.

No.	Reference letter of Maker.	*a*	*b*	*c*	*d*	*e*	*f*	*g*
1	Pressure on pads—lb. per sq. in. Normal Maximum	Up to 150 350	*	Up to 250 300	Up to 150 500	Up to 300 500	Up to 150 500	Up to 200 400
2	Surface velocity V at mean dia., ft. per sec. Normal Max.	50 - 120 187	*	130 - 170 216	150 172	60 - 140 150	100 190	120 160
3	Ratio of breadth to length of pad, B/L	1/1	*	1/1	1/1	1/0·93 1/1·15	1·1/1·0	1/1·7
4	Ratio of thickness to length of pad, T/L ...	1/3	*	1/2·0 to 1/2·5	1/3	1/2·48 to 1/3·75	1/3	1/4
5	Minimum thickness of white metal, inches, *t*	1/16	*	1/16	1/16	3/32	1/8	0·06
6	Method of finishing pad white metal surfaces ...	Scraping.	*	Scraping.	Scraping.	Scraping.	Scraping.	Scraping.
7	Percentage composition of white metal used ...	Sn 86 Sb 8·5 Cu 5·5	*	Sn 76 Sb 10 Cu 6 Pb 8	Sn 83·3 Sb 8·33 Cu 8·33	Sn 83 Sb 9 Cu 8	Sn 77 Sb 14 Cu 9	Sn 85 Sb 8·5 Cu 6·5
8	Amount of eccentricity *e*/L of pivots from centre line of pads	10 per cent. $=L/10$ $a/b=2/3$	*	16 per cent. $=L/6{\cdot}25$ $a/b=1/1{\cdot}94$	5 per cent. $=L/20$ $a/b=9/11$	16·7 p. cent. $=L/6$ $a/b=1/2$	10 per cent. $=L/10$ $a/b=2/3$	Zero
9	Total oil clearance, mils. Minimum Normal Maximum	12 14 20	*	8 — 12	$0{\cdot}82\sqrt{VL}$ — —	14 — 40	4 to 5 per cnt. 1 in. of L —	10 — 12
10	Quantity of oil in cub. ft. per min. is made sufficient for	20 deg. F. temp. rise (from 12)	*	25 deg. F. temp. rise (from 12)	15 deg. F. to 25 deg. F. (from 12)	—	15 deg. F. temp. rise (from 12)	15 deg. F. to 25 deg. F. (from 12)

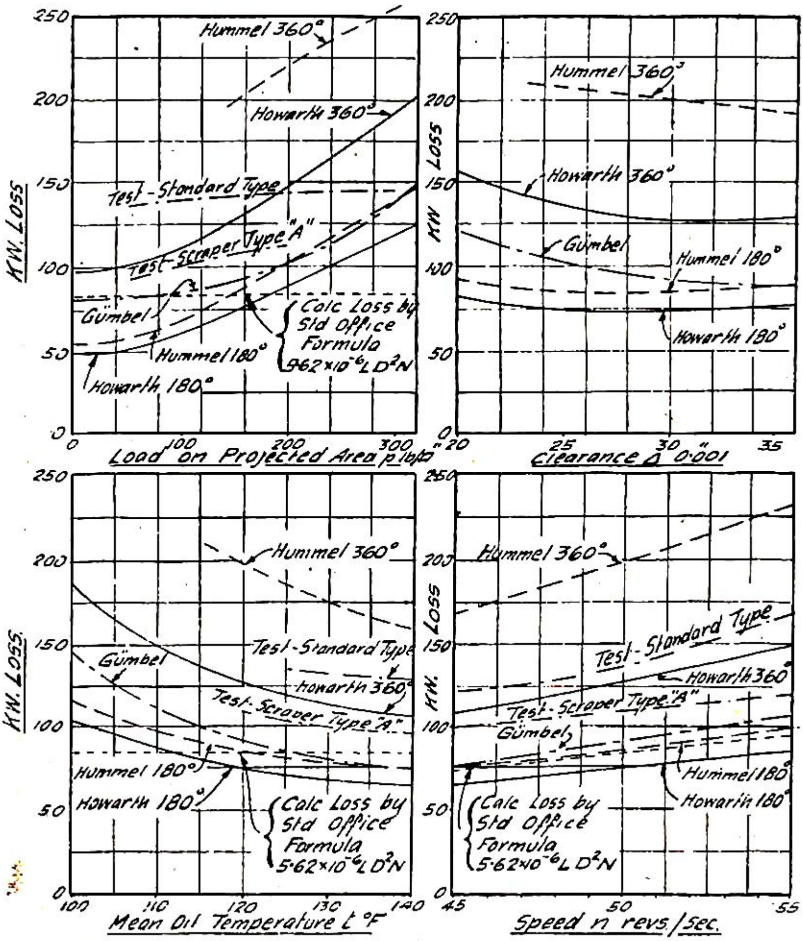
APPENDIX 2. Taken from **TABLE 1**—Turbine Journal Practice, by H. L. GUY and D. M. SMITH. General Discussion on Lubrication and Lubricants. I.Mech.E., 1937.

MANUFACTURER.	A	B	C	D	E	F
Load on projected area, lb. per sq. in.						
Minimum, ...	50	45	30	27	30	—
Normal, ...	—	100 to 150	100	80 to 130	70 to 120	57 to 114
Maximum (practice) ...	146	160	130	170	150	200
Surface speed, ft. per sec. :						
Normal	150	Up to 150	130	Up to 157	Up to 144	—
Maximum (practice) ...	170·5	177	160	170	157	165
Ratio, length/diameter :						
Minimum	0·9	0·8	1	0·75	1	1
Normal	1 to 1·5	1	1·25	1 to 1·5	1·25	1·6
Maximum	2·5	1·5	2·0	2	1·5	—
Clearance per inch diameter, ins.	0·0015 to 0·002	0·002	0·0015	0·002 to 0·003	0·001 to 0·002	Vertical, 0·001 Horizntl., 0·004
Minimum thickness of white metal journal, diam. 4 ins., ins.	0·16	3/16	3/16	1/4	5/16	0·2
Minimum thickness of white metal journal, diam. 12 ins., ins.	0·28	3/8	1/4	3/8	3/8	0·5
Percentage composition of white metal : Tin ...	76	83·4	82·9	86 72-76	77	86 78
Antimony	10	8·3	8·3	8·5 6-7	14	8·5 13
Copper	6	8·3	8·3	5·5 4-5	9	5·5 6
Lead ...	8	—	0·5	— 13·5-16·5	—	— 3
Approximate oil quantity, gal. per min. :						
10×10 in. bearing 3000 r.p.m.	16	26·7	11·6	11·3	13·7 to 20·6	—
20×20 in. bearing 1500 r.p.m.	48	75	46·4	45·2	54 to 82	—

APPENDIX 3. Taken from BAUMANN, K. "Some Recent Developments in Power Station Practice." Inst. of Eng., Australia, 1938.

15" × 22" EXPERIMENTAL JOURNAL BEARING

CALCULATED AND TEST LOSSES



Std Conditions:- p = 150 lb/□", clearance = 0·030", oil temp. = 120°F., speed = 50 revs/sec

Test Bearing Losses plotted for 40 G.P.M.

APPENDIX 4.

Values taken from BOSWALL, R. O. "The Film Lubrication of the Journal Bearing." Proc. Inst. of Mech. Engineers, 1932. Part 1, Vol. 122.

Arc Sub-tended	Type of Loading.								
	0·6 Eccentric.			Central.			0·4 Eccentric.		
	K	m	C	K	m	C	K	m	C
45°	0·0026	0·62	0·00047	0·0028	0·58	0·00050	0·0029	0·58	0·00049
60°	—	0·62	0·00070	—	0·58	0·00060	—	0·58	0·00052
75°	—	0·62	0·00092	—	0·58	0·00072	—	0·58	0·00052
90°	0·0024	0·62	0·00108	0·0028	0·58	0·00080	0·0032	0·58	0·00048
105°	—	0·62	0·00110	—	0·58	0·00082	—	0·58	0·00036

APPENDIX 5.

Information abstracted from the MINISTRY OF SUPPLY. Publication P.B.I., 1942.

(2) Recommendations involving alterations in practice or design.

(a) **White Metal.**

(i) Reduction in thickness of white metal . . . .

It is known that pre-finished and ready-to-fit bearings, designed according to the following table, are performing satisfactorily :—

Bearings below	2½" bore,	Linings,	·030" thick.
,,	2½" - 4"	,,	·040"
,,	4" - 6"	,,	·060"

The thickness of white metal in unflanged, thin-walled bearings, should be reduced to ·010"/·015"

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2. Deflection of Shafts and Beams
3. Deflection of Shafts and Beams (Instruction Sheet) } Connected.
4. Steam Radiation Heating Chart.
5. Horse-Power of Leather Belts, etc.
6. Automobile Brakes (Axle Brakes)
7. Automobile Brakes (Transmission Brakes) } Connected.
8. Capacities of Bucket Elevators.
9. Valley Angle Chart for Hoppers and Chutes.
10. Shafts up to 5½-in. diameter, subjected to Twisting and Combined Bending and Twisting.
11. Shafts, 5¾ to 26-inch diameter, subjected to Twisting and Combined Bending and Twisting.
12. Ship Derrick Booms.
13. Spiral Springs (Diameter of Rd. or Sq. Wire).
14. Spiral Springs (Compression).
15. Automobile Clutches (Cone Clutches).
16. ,, ,, (Plate Clutches).
17. Coil Friction for Belts, etc.
18. Internal Expanding Brakes. Self-Balancing Brake Shoes (Force Diagram).
19. Internal Expanding Brakes. Angular Proportions for Self-Balancing. } Connected.
20. Referred Mean Pressure Cut-Off, etc.
21. Particulars for Balata Belt Drives.
22. ⅞″ Square Duralumin Tubes as Struts.
23. 1″ ,, ,, ,, ,,
24. ¾″ Sq. Steel Tubes as Struts (30 ton yield).
25. ⅞″ ,, ,, ,, (30 ,,).
26. 1″ ,, ,, ,, (30 ,,).
27. ¾″ ,, ,, ,, (40 ,,).
28. ⅞″ ,, ,, ,, (40 ,,).
29. 1″ ,, ,, ,, (40 ,,).
30. Moments of Inertia of Built-up Sections (Tables)
31. Moments of Inertia of Built-up Sections (Instructions) and Examples). } Connected.
32. Reinforced Concrete Slabs (Line Chart)
33. Reinforced Concrete Slabs (Instructions and Examples) } Connected.
34. Capacity and Speed Chart for Troughed Band Conveyors.
35. Screw Propeller Design (Sheet 1, Diameter Chart)
36. ,, ,, ,, (Sheet 2, Pitch Chart) } Connected.
37. ,, ,, ,, (Sheet 3, Notes and Examples)
38. Open Coil Conical Springs.
39. Close Coil ,, ,,
40. Trajectory Described by Belt Conveyors.
41. Metric Equivalents.
42. Useful Conversion Factors.
43. Torsion of Non-Circular Shafts.
44. Railway Vehicles on Curves.
45. Chart of R.S. Angle Purlins.
46. Coned Plate Development.
47. Solution of Triangles (Sheet 1, Right Angles).
48. Solution of Triangles (Sheet 2, Oblique Angles).

49. Relation between Length, Linear Movement and Angular Movement of Lever. (Diagram and Notes).
50. ,, ,, ,, ,, ,, ,, (Chart).
51. Helix Angle and Efficiency of Screws and Worms.
52. Approximate Radius of Gyration of Various Sections.
53. Helical Spring Graphs (Round Wire)
54. ,, ,, ,, (Round Wire) } Connected.
55. ,, ,, ,, (Square Wire)
56. Relative Values of Welds to Rivets.
57. Ratio of Length/Depth of Girders for Stiffness.
58. Graphs for Strength of Rectangular Flat Plates of Uniform Thickness.
59. Graphs for Deflections of Rectangular Flat Plates of Uniform Thickness.
60. Moment of Resistance of Reinforced Concrete Beams.
61. Deflection of Leaf Spring.
62. Strength of Leaf Spring.
63. Chart showing relationship of various Hardness Tests.
64. Shaft Horse Power and Proportions of Worm Gears.
65. Ring with Uniform Internal Load (Tangential Strain) } Connected.
66. Ring with Uniform Internal Load (Tangential Stress)
67. Hub Pressed on to Steel Shaft. (Maximum Tangential Stress at Bore of Hub).
68. Hub Pressed on to Steel Shaft. (Radial Gripping Pressure between Hub and Shaft).
69. Rotating Disc (Steel) Tangential Strain } Connected.
70. ,, ,, ,, ,, Stress
71. Ring with Uniform External Load, Tangential Strain } Connected.
72. ,, ,, ,, ,, ,, Stress
73. Viscosity Temperature Chart for Converting Commercial to Absolute Viscosities.
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75. Ring Oil Bearings.
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81. Involute Toothed Gearing Chart.
82. Steam Pipe Design. Chart showing flow of Steam through Pipes.
83. Variation of Suction Lift and Temperature for Centrifugal Pumps.
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86. ,, ,, ,, ,, ,, } Connected.
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